A tale of two motors

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In 1992, a large refinery purchased two 30,000 hp, four pole, synchronous motors for a new Fluidized Catalytic Cracker (FCC) unit. One motor was to drive a low pressure, high volume, axial flow compressor through a speed increasing gearbox (Figure 1). This compressor supplied air to the FCC reactor and, therefore, was a critical machine for the entire FCC unit. The second motor was purchased as a spare because of the cost of lost production time and the long lead times for a replacement if something happened to the first motor. Motor #1 started operation in 1993 and ran well until the middle of 1994, when shaft and bearing housing vibration amplitudes began to slowly increase. By the beginning of 1995, the vibration amplitude had risen to a level close to the machinery protection systems alarm points, and the plant's Predictive Maintenance personnel contacted Bently Nevada Machinery Management Services, the OEM, and other machinery consultants, to help diagnose the problem.

Initially, only steady state data could be collected because of the need to continue unit operation. This data indicated a high 1X amplitude on the rotor and bearing housings. The bearing pedestals were extremely compliant, and amplitudes from 80% to 130% of the shaft relative readings were typical. Compared to the steady state data from the plant's extensive history on the unit, the relative and seismic 1X vibration amplitudes were larger and still increasing. Long-term increases in the shaft 1X vibration are disturbing because they may be an indicator of shaft crack propagation. The first rule of shaft crack diagnosis states that if a rotor is cracked, it is almost certainly bowed. The bowing and the lower rotor stiffness leads to an increase in the rotor 1X vibration for a unit operating below its first balance resonance, as this unit did. But the confirming evidence of a rotor crack could only come from transient data taken during startup or shutdown, and the next outage was not scheduled until the summer of 1996.

Other common causes of an increase in 1X amplitude are a decrease in the rotor support system stiffness (bearing or foundation), an increase in the unbalance, or the introduction of a rub-induced thermal bow. In long-term trends, increasing unbalance and decreasing system stiffness, generally as a result of foundation degradation, are the most common problems.

Some first aid

All of the motor's seals were inspected for evidence of a stationary part rub. The primary frequency component of 1X could have been evidence of a thermal bow induced by a light rub. No evidence of seal rubs was discovered, and examination of the motor foundation revealed an unusual support structure for a machine this large (Figure 2). The motor base frame was only supported by 4 x 4 inch shims under each hold-down bolt, and the space between the foundation bedplate

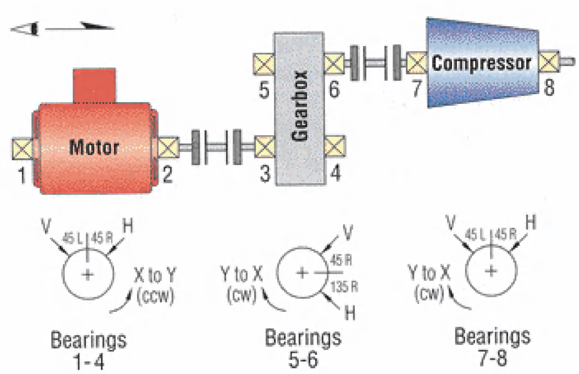

Figure 1. Synchronous motor-driven compressor train for Fluidized Catalytic Cracker unit.

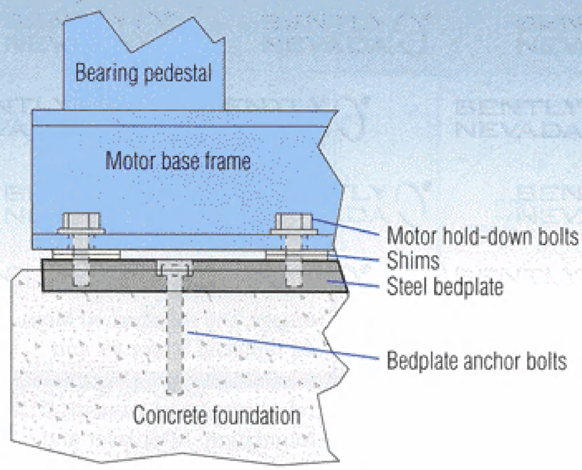

Figure 2. Mounting detail of original motor foundation. Frame supported only by shims and not grouted to bedplate.

and the frame was not grouted. To increase the support stiffness, and as a temporary solution until a better foundation modification could be implemented, double-tapered wedges were driven between the motor frame and the foundation bedplates at several locations. The hold-down bolts were re-torqued. The immediate effect this increase in the foundation stiffness had on the operational amplitudes was to reduce the 1X vibration amplitudes by over 50% (Figure 3). A foundation upgrade was planned for the 1996 outage, and the wedges were re-tightened several times during the year when the 1X vibration amplitudes began to increase.

More of the puzzle

In the summer of 1995, the unit was shut down because of a hurricane in the area. Transient data was acquired during the outage to check for further evidence of shaft cracking and to determine if the temporary wedges needed re-tightening. Shortly after startup, transient and steady state data revealed an alarming change in the slow roll vector and a large shift in the 1X phase (both are possible indicators of shaft cracking). However, data sampled during the next two days, as the compressor was loaded to full capacity and the motor current increased to maximum, indicated a large thermal vector change in the rotor response (Figure 4). Although amplitudes remained fairly consistent, the 1X phase shifted over 90° on both bearings. As the rotor temperatures stabilized, the 1X amplitudes and phases trended back to levels recorded prior to the shutdown. The rotor contained a very large, previously unknown thermal vector. This thermal vector was responsible for the difference in the slow roll values, as well as the difference in the 1X amplitude and phase from hot to cold.

A short lesson in thermal changes

Thermal vectors in rotors, especially large motors and generators, are not uncommon. In most cases, they are the result of electrical faults (turn-to-turn or turn-to-rotor shorts). The resultant I^2R (resistive) losses at the short cause localized heating of the rotor, resulting in a bow due to the local expansion of the rotor iron. The bow changes the balance condition of the rotor, thus changing the 1X vibration amplitude (up or down) and phase. Except for the loss in efficiency of the motor or generator, one or two shorted turns do not degrade the mechanical integrity of the machine. If not severe, the bow is generally offset with balance weights until the problem can be corrected. Another cause of thermal rotor bows is nonhomogeneity of material in the rotor forging. When the rotor blank is heat treated and hardened, a nonuniform hardness in the forging will cause the rotor to bow when heated in operation. Inclusions in the material may also cause a rotor bow.

Although not often encountered, generator rotors have developed thermal bows because of nonuniform tightness of the windings. As the windings heat, they expand axially with the rotor. If one or more sets of

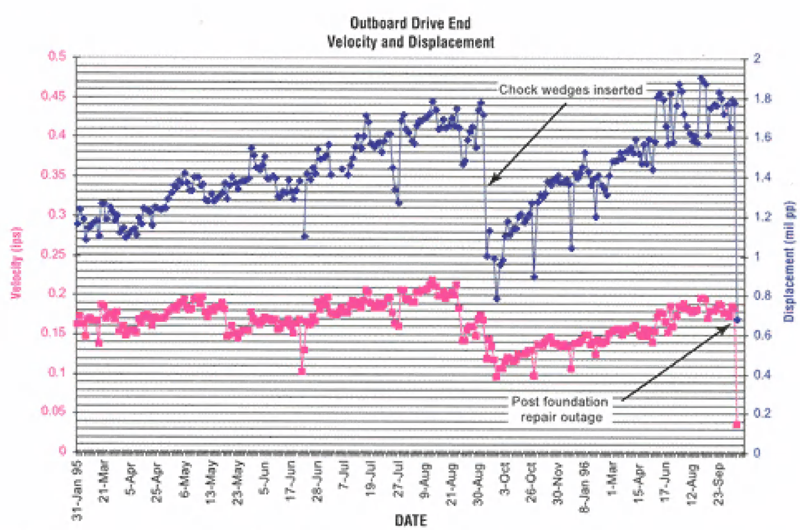

Figure 3. Effect of increased mounting stiffness on shaft relative (displacement) and housing (velocity) vibration.

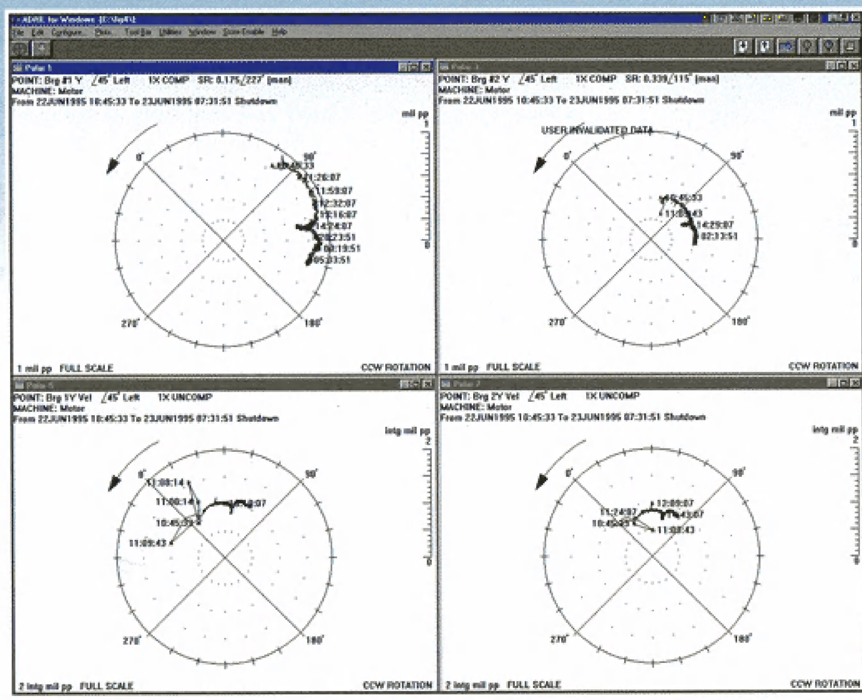

Figure 4. Motor #1 thermal vector change in rotor response over 21 hours as compressor was loaded to full capacity.

windings are constrained by tight wedges, tremendous forces are built up which will bow the rotor. (Note: this type of thermal bow can be a sudden step change rather than a slow movement. See "Shaft crack detected at an ammonia plant," September 1992 Orbit). Thermal vectors from most motor/generator electrical and material nonhomogeneity problems can be difficult to diagnose. However, even though they are caused by an unknown "problem" within the rotor, their direction and amplitude are repeatable and often can be controlled with load. The vector is also independent of the initial balance condition. It is this repeatability and independence that allow balance correction weights to be a very effective way to reduce the effects of a thermal vector. The hot operational amplitudes are balanced to achieve low running amplitudes without excessively high cold starting amplitudes. If the thermal vector is large, there must often be a compromise in the cold and hot operational amplitudes. (Note: while it is common industry practice to correct rotor vibration amplitudes in this manner, large thermal vectors, offset by balance weights, may be a symptom of a problem that should be corrected, not masked.)

Motor #2

In the fall of 1996, a major outage on the FCC unit allowed Motor #1 to be removed for foundation upgrades. Prior to this, Motor #2, in storage for three years, was returned to the OEM for upgrades to stiffen its base frame, a complete electrical check, cleaning, and insulation modifications. The additional 8,000 horsepower that resulted from insulation upgrades would allow the plant to produce more product. During the outage, the refurbished Motor #2 replaced Motor #1 and was installed on a completely refurbished and grouted foundation. After laser and optical alignment checks were completed, a sixteen hour heat soak at full load (a 30,100 lb rotor takes awhile to heat soak) showed that this motor also had a thermal vector (Figure 5). Two heat soak runs proved the repeatability in phase and magnitude of this vector, and the rotor was trim balanced for full load operation at close to 1,100 stator amps. During the outage, several more trim balance runs reduced the operational amplitudes to extremely low levels, and again proved the repeatability of the thermal vector.

Over the next eight months, plant requirements dictated that the compressor train operate at less than the expected load of 1,100 stator amps. As expected, vibration amplitudes began to increase as motor operating temperatures and the thermal vector magnitude decreased. The 1X vibration amplitude and phase approximately followed the change in the thermal vector toward the cooler operating condition.

In September, 1997, when the FCC unit was shut down for reactor problems, the motor was trim balanced for the lower operating temperatures. To confirm repeatability of the thermal vector, the unit was heat soaked without any change to its balance condition.

Surprises

Figure 6 shows a comparison of three heat soak runs on Motor #2 on 2 December, 1996 and 4 and 5 September, 1997. For clarity, the initial cold starting point of the thermal vector has been moved to the center of the polar plot. As you can see, the magnitude and direction of the vector was not repeatable. The runs on 2 December and 4 September are with no change to the balance condition. The run on 5 September was an

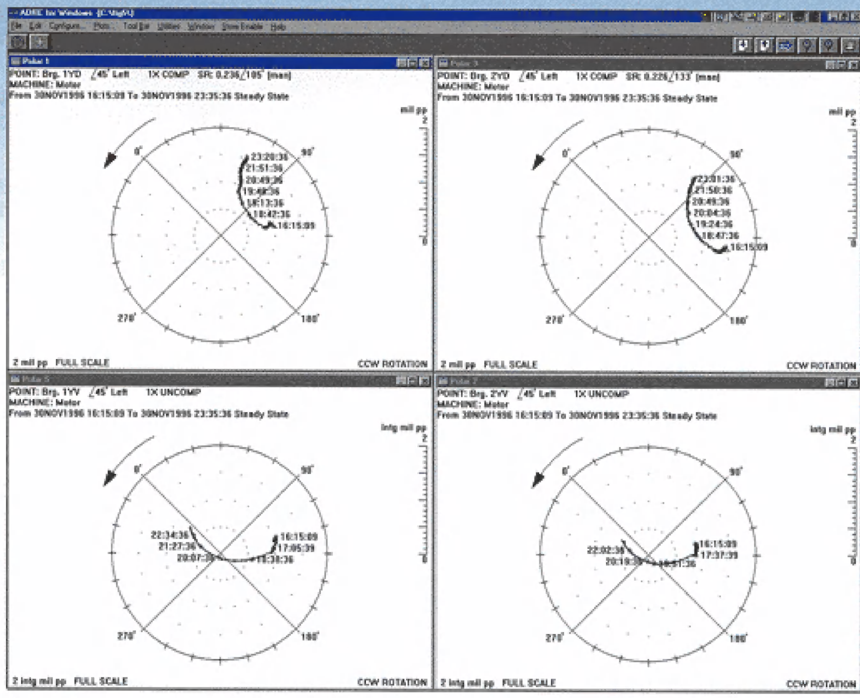

Figure 5. Motor #2 thermal vector change in rotor response during part of heat soak cycle at full load.

attempt to balance the 4 September thermal vector. This proved to be an impossible trim balancing problem. Although the rotor responded well to initial balance weights, and the balance influence data was consistent, the changing direction and length of the heat soak thermal vector made the operational amplitude and phase unpredictable.

The inability to balance the rotor, and concerns of a possible rotor crack, loose windings, or unknown electrical problem, led plant personnel to the correct decision to replace Motor #2 with Motor #1. The thermal vector in Motor #1 was trim balanced, and the unit is in operation. However, there are indications of a slight upward trend in vibration amplitude which plant personnel are monitoring very closely. Motor #2 was returned to the OEM for evaluation and repair. Seven heat soak runs on the OEM test floor produced the thermal vectors shown in Figure 7. Again, for clarity, the initial cold starting point of the thermal vector has been moved to the center of the plot.

Summary

Most thermally-sensitive rotors are very predictable, well-behaved units that have straightforward root causes for their behavior. The fact that the magnitude and direction of the vector were not repeatable was extremely unusual. Teardown of the motor revealed loose copper windings on all four poles. Jacking bolts on the end turns of one pole were loose, and cracked insulation between the windings and pole tips was also discovered. The post overhaul testing and balancing of the motor returned a repeatable thermal vector and a predictable balance response. While the exact cause of the large thermal sensitivity and its lack of repeatability remain unknown, loose windings are suspected to be the most likely cause. Differential tightness of each pole winding set will produce differential axial strain on the rotor as the windings expand axially.

The principal reasons for thermal sensitivity in large motors and generators are electrical problems in the form of turn-to-turn or turn-to-rotor shorts. There are a variety of electrical tests (surge comparisons, resistance to ground, high potential, current balance analysis, current signature analysis, and several others) that will diagnose these problems. However, most tests require either the unit to be out of normal service or specialized test equipment. Vibration characteristics can help verify a thermally-sensitive rotor and the root cause of the sensitivity while the unit is in operation.

Typically, thermal sensitivity is characterized by a "slow" change in the 1X steady state (constant rpm) vibration amplitude and/or phase as the rotor heats or the load on the unit increases. The slow roll will also change between the cold and hot condition. When the rotor is bowed, proximity readings will differ from hot to cold during slow roll. In most cases, the magnitude and direction of the thermal change vector will be independent of the initial balance condition. Also, in most cases, the magnitude and direction from hot to cold will be repeatable and often "controllable" (in electrical machinery) with unit load or power factor adjustments. In generators, a high reactance load (vars) will increase the localized heating and help verify the presence of shorted turns. In large synchronous motors, lowering the power factor has the same effect.

Thermal bows encountered in nonelectrical machinery, such as pumps, compressors, and turbines, are usu-

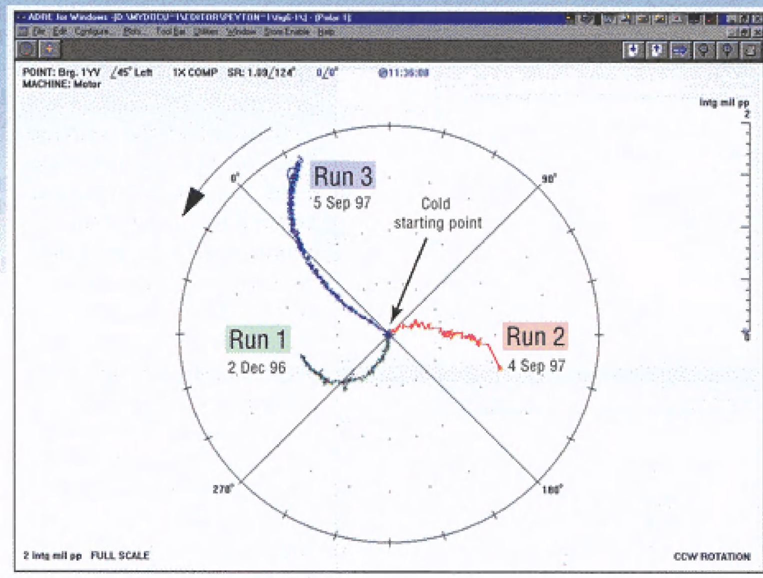

Figure 6. Motor #2 thermal vector changes. Run 1 and run 2 are with no change in balance condition. Run 3 shows results of trim balancing attempt using run 2 data.

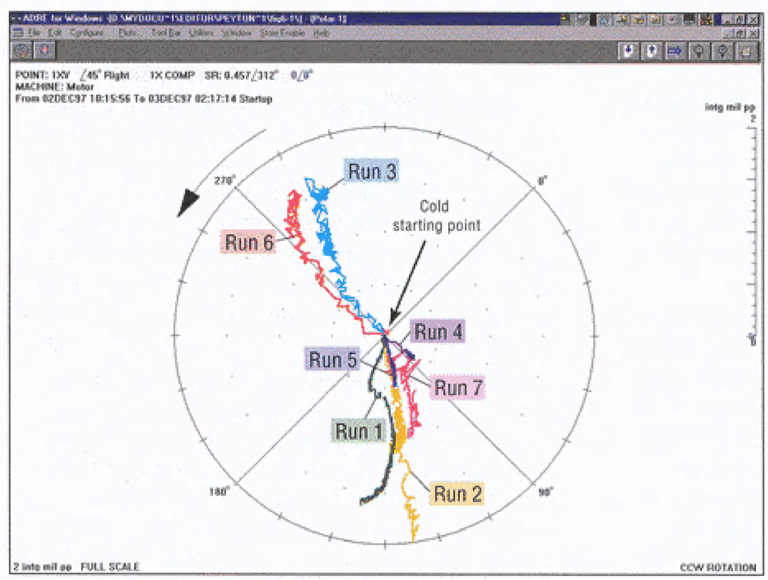

Figure 7. Motor #2 thermal vector changes from seven heat soak runs performed on OEM test floor.

ally the result of rubs or of material nonuniformity. The nonuniformity is due to improper heat treatment or annealing and effects of localized welding or machining. The bows occur as the process fluid heats the rotor and are more challenging to identify because the rotor temperatures are not easily varied or controlled through a wide range.

Thermal bows produce a change in the rotor balance condition and, thus, a change in the 1X vibration amplitude and phase. Other frequency components should remain unaffected by the bow unless the bow is severe enough to create other problems, such as a rub.

Operational vibration amplitudes can effectively be reduced in the short term by trim balancing; however, the root cause of the problem should be addressed during the next outage. Remember, you are "correcting" a rotor bending force with a mass unbalance force. Large bows "balanced" with large correction weights can place high bending forces on the rotor. Vibration amplitudes through transients can also be severe. In variable speed machinery, such as motors equipped with variable frequency drives, balancing will be a compromise. Confirmation of a machinery problem should always include a thorough examination of steady state and transient data in a variety of formats (including Bode, polar, and full spectrum cascade plots). Use all the tools that are available to you.

The value of a good Predictive Maintenance program, based on frequent machine sampling or continuous online vibration monitoring, cannot be overemphasized. This plant has an extensive and well-run predictive maintenance program that has not only saved tremendous downtime production costs, but has also contributed to the safer operation of the plant and fewer catastrophic machine failures.

All too often, an increase in the 1X vibration is assumed to be a change in the "balance" condition of machinery, and is "corrected" with trim balance weights without regard to the root cause of the change. When the operation of machinery changes, don't rush to fix the symptom without understanding the problem. Masking symptoms can lead to catastrophic failure.